

Experimental investigation of a free-piston Vuilleumier refrigerator

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This paper presents experimental data for a free-piston Vuilleumier machine, which was designed as a refrigerator for CFC-free cooling and air conditioning. The basic features of the Vuilleumier cycle and the free-piston operation, which is known from free-piston Stirling engines, are discussed briefly. Additionally, the design of the prototype is described. From the variety of the experimental data, the influence of the mean pressure and the three cycle temperatures on the refrigerating capacity and the COP are selected for presentation and detailed discussion in this paper. The results are used to deduce some general operating characteristics of a free-piston Vuilleumier refrigerator, which reveal its potential for CFC-free cooling and air conditioning.

(Keywords: refrigeration cycle; heat pump; Vuilleumier; free piston; prototype; performance)

Recherche expérimentale sur un réfrigérateur de type Vuilleumier à piston libre

L'article présente les données expérimentales d'une machine de type Vuilleumier à piston libre qui a été conçue pour le refroidissement et le conditionnement d'air sans CFC. On examine brièvement les caractéristiques de base d'un cycle Vuilleumier et le fonctionnement d'un piston libre, que l'on retrouve dans les moteurs de Stirling à piston libre. En outre, on décrit la conception d'un prototype. Parmi les nombreuses données expérimentales, on a choisi celles qui caractérisent l'influence de la pression moyenne et des trois températures du cycle sur la capacité de refroidissement et le COP, afin de les présenter et de les examiner dans cet article. D'après les résultats, on obtient quelques caractéristiques générales du fonctionnement d'une machine de type Vuilleumier à piston libre; elles révèlent les possibilités de cette machine à être utilisée en refroidissement et conditionnement d'air sans CFC.

(Mots clés: cycle frigorifique; pompe à chaleur; Vuilleumier; piston libre; prototype; performance)

Owing to the demand for CFC-free cooling and air conditioning, the Vuilleumier cycle, which was patented by R. Vuilleumier in 1918¹, is under discussion as a new refrigeration technique. Hydrogen, helium or air, which are the preferred working gases, have no environmental impact at all.

In recent years, various Vuilleumier machines have been developed both for heating and cooling²⁻⁴, but owing to their low power-to-weight ratios they are not competitive yet. In order to improve this ratio, the authors propose the design of a Vuilleumier machine with free pistons⁵. In free-piston machines the cyclic variation of the working gas pressure is used to excite the oscillating motion of the displacers. Thus there is no need for a crank mechanism, resulting in a reduction of size, weight and complexity of the machine. Free-piston Stirling

machines have been under investigation for more than 20 years, and their characteristics are generally known and understood⁶. The suitability of a Vuilleumier machine for free-piston operation was proved by a prototype built at the University of Dortmund. The experimental data for this machine will be presented in the following.

The Vuilleumier cycle

The Vuilleumier cycle is a closed cycle. Its working gas is transferred between three volumes each of a characteristic temperature level: cold, warm and hot. *Figure 1* shows a schematic Vuilleumier machine. It consists of a cylinder with a cold (1) and a hot displacer (2), which divide the total volume in three spaces. These

Nomenclature

A_r	Displacer rod area	x	Position
COP	Coefficient of performance (refrigerating capacity/hot heat input)	$\dot{x}$	Velocity
$\text{COP}_{\text{Carnot}}$	COP of the Carnot cycle at corresponding temperatures	$\ddot{x}$	Acceleration
D	Coefficient of damping	ω	Frequency
F_C	Force exerted by the spring C	Subscripts	
K	Spring constant	c	Cold
m	Displacer mass	h	Hot
p	Working gas pressure	w	Warm
p_B	Gas spring pressure	1	Cold displacer
T	Temperature	2	Hot displacer

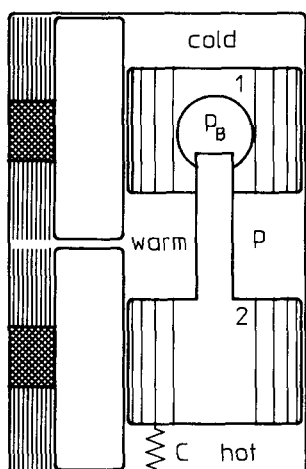


Figure 1 Configuration of a free-piston Vuilleumier machine
 Figure 1 Configuration d'une machine de type Vuilleumier à piston libre

spaces are kept at the temperature levels required by the adjacent heat exchangers. Additionally, there are two regenerators located at the interfaces where the volumes of the different temperature levels meet. They work as heat stores and guarantee a suitable temperature of the working gas entering the cylinder spaces.

It is obvious that the pressure is spatially constant, if the pressure drop in the heat exchangers and regenerators is neglected. However, pressure varies in time owing to the thermal compression and expansion as the working gas is transferred to different cylinder spaces. Tuning the displacers in a way that the motion of the hot displacer lags behind the cold one, the compression arises when the gas is mainly in the warm cylinder space. Thus the heat of compression can be removed from the cycle at the warm temperature level. Evidently, the cold and the hot space contain most of the gas during the expansion, requiring an appropriate heat supply at the cold and hot temperature level.

Mechanical compression is created by the displacer rod only, but the effect on the cyclic pressure variation is small compared with the thermal compression. However, the displacer rod acts like a small power piston producing pV work, which is needed to compensate for the power losses due to mechanical friction and viscous

dissipation. Thus there is no need for any external supply of mechanical power and a Vuilleumier machine can be termed as a self-sustaining, thermally activated heat pump or refrigerator.

The free-piston Vuilleumier machine

In free-piston machines the cyclic variation of the working gas pressure is used to move the displacers. This offers some important advantages compared with crank-driven machines:

1. There is no need to transform the reciprocating motion of the displacers into a circular motion. Thus the side loads on the moving parts are negligible, which reduces friction and wear and allows a long operating life and low maintenance.
2. All forces transferred from the machine to the frame are directed axially. Thus they can easily be absorbed by a linear damper, resulting in an extremely quiet operation with little vibration.
3. There are only two moving parts, which improves the reliability and reduces the weight, size and manufacturing cost of the machine.

Figure 1 shows that the displacers are suspended by springs. While the hot displacer is connected to the casing by spring C, the cold displacer is connected to the hot displacer by a gas spring formed by its internal volume. Thus each displacer forms a potential oscillator with a resonant frequency, given by

$$\omega = \sqrt{\frac{K}{m}}$$

K represents the resulting constant of the springs acting on the displacer, and m is its mass. In order to maintain stationary oscillation, the pressure difference between the working space and the gas spring across the displacer rod area is used as exciting force. This is illustrated in Figures 2 and 3.

The phasor diagram in Figure 2 represents the phases of the displacer motions and the pressure waves. The phase angle between the two displacers is 90° , which is the optimum value for the ideal Vuilleumier cycle in view of refrigerating capacity. The working gas pressure p is

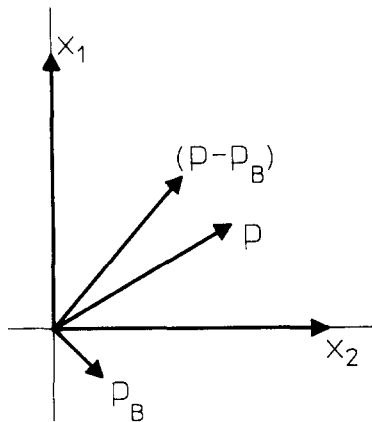


Figure 2 Phasor diagram for the displacer motions and the pressure waves

Figure 2 Diagramme de phase pour les mouvements du piston et les ondes de pression

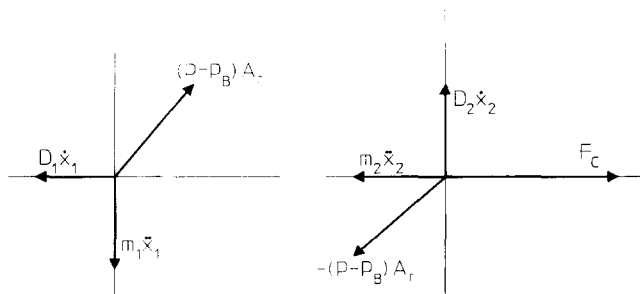


Figure 3 Force diagrams for the displacers

Figure 3 Diagrammes de la force des pistons

almost in phase with the motion of the hot displacer. If the hot displacer is near its upper dead position, most of the gas is located in the hot cylinder space, causing a pressure maximum (Figure 1). In real machines the working gas pressure precedes the motion of the hot displacer by a phase angle of approximately 30° . The phase shift of the gas spring pressure p_B can be calculated directly from the displacer motions, and the pressure wave lags behind the motion of the hot displacer by 45° , if the displacer amplitudes are equal. In order to determine the phase shift of the pressure difference $(p - p_B)$, it is assumed that the amplitude of the gas spring pressure is approximately three times smaller than the amplitude of the working gas pressure.

A necessary condition for oscillation is an equilibrium of the inertial force, the damping forces and the external forces for each displacer, which is presented by the vector diagrams in Figure 3. Here, the damping due to viscous coupling was neglected, since these forces are one order of magnitude smaller than the damping forces created by the displacer itself⁷. In the case of the cold displacer the only external force is formed by the pressure difference $(p - p_B)$ across the displacer rod area. Keeping in mind that the velocity and acceleration are preceding the position vector by 90° and 180° , the damping and the inertial force can be plotted in the vector diagram, and it can be seen that equilibrium is achieved. Looking at the hot displacer, it can be seen that the pressure difference

force is 180° out of phase, and a second external force is exerted by the spring C. Obviously, the superposition of the external forces compensates for the damping and inertial force, and equilibrium is achieved as well. This illustrates the suitability of a free-piston Vuilleumier machine for self-excited, harmonic oscillation.

Examining the undamped oscillation system shows the existence of exactly one operating frequency, which can be approximated by the arithmetic mean of the natural frequencies of the displacers. The damped system can be evaluated using a linear model. This is advantageous, because the linear model allows closed-form solutions to find the operating frequency, the phase shift between the displacers, the ratio of the displacer amplitudes and one additional parameter (i.e. a displacer mass)⁸. However, the non-linearities within the system, such as turbulent flow losses, have to be taken into consideration, as they guarantee stability of operation. The investigation of the non-linear system also shows that the slightest perturbation is sufficient for starting, if sliding friction is zero. In real machines a certain starting amplitude has to be exceeded to overcome the sliding friction. After starting, the displacer amplitudes increase until the stable point of operation is reached.

The prototype

A first prototype of a free-piston Vuilleumier machine was built at the University of Dortmund. It should be emphasized that this prototype was designed to prove the feasibility of a free-piston Vuilleumier machine and to study its operational characteristics. High performance in terms of refrigerating capacity and COP was not the major design goal. Instead, a design allowing variation of the main parameters, such as spring constants and displacer masses, in a wide range was much more important. The basic design parameters of the prototype are listed in Table 1.

Figure 4 shows the basic design features of the prototype. The main pressure vessel is formed by two cylinders, which are flanged together and sealed with an O-ring. In order to supply and remove the heat from the cycle, the cylinders are covered with water jackets in the region of warm temperature and with an ethanol jacket in the cold region. The hot end is shrunk into a steel block, which is heated electrically. The internal heat

Table 1 Basic design parameters of the prototype

Tableau 1 Paramètres de base de la conception du prototype

Working gas	Helium
Hot temperature (wall)	500°C
Warm temperature (water circuit)	20°C
Cold temperature (coolant circuit)	0°C
Mean pressure	20 bar
Cylinder bore	66 mm
Desired dynamic parameters:	
Frequency	10 Hz
Displacer phase	90°
Displacer strokes	30 mm each
Approx. performance	
Refrigerating capacity	200 W
COP ^a	0.7

^aThe COP of a free-piston Vuilleumier refrigerator can be evaluated from the ratio of the refrigerating capacity to the heat input at hot temperature level

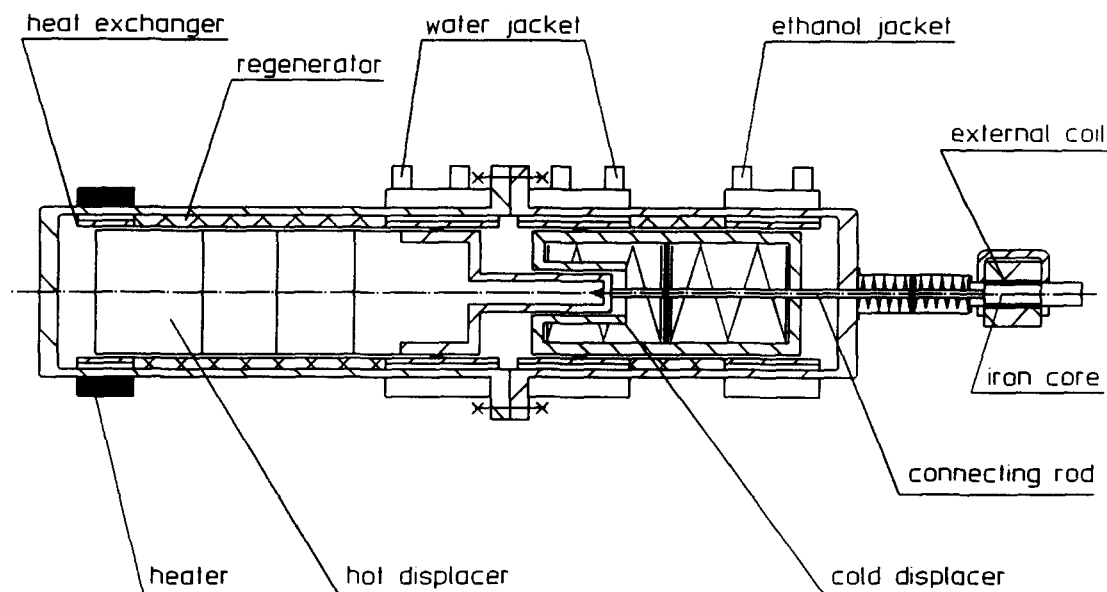


Figure 4 Design sketch of the prototype

Figure 4 Description de la construction du prototype

Table 2 Experimental data at the design point

Tableau 2 Données expérimentales au point de conception

Frequency	10.02 Hz
Displacer phase	84.1°
Displacer stroke (cold)	28.0 mm
Displacer stroke (hot)	29.4 mm
Stroke ratio	0.952
Refrigerating capacity	71 W
COP	0.11

exchangers are concentric steel rings with fins on the outside. They are pressed into the cylinders to achieve a sufficient thermal contact with the cylinder wall. Both regenerators are made of stainless steel foil, which is corrugated, providing several triangular flow passages for the working gas. The arrangement of the displacers and the springs can also be seen in Figure 4. The displacers run on the inside of the two warm internal heat exchangers. Instead of guide rings, the running surfaces are coated with a PTFE-based material to minimize friction and wear. The gas spring within the cold displacer is supported by a pair of mechanical springs. This is advantageous, because a defined rest position of the cold displacer can be maintained and gas spring hysteresis losses are reduced as well as non-linearities. The spring connecting the hot displacer with the casing (corresponding to spring C in Figure 1) is located in a separate casing outside the main pressure vessel. The connection is formed by a thin rod, guided through the pressure vessel and the cold displacer. The outer end of this rod is attached to an iron core, which can be deflected by an external coil to start the machine.

The length of the main pressure vessel is 435 mm, the outer diameter of the flange is 120 mm and the machine weights approximately 16 kg including the electric heater.

Experimental results

Table 2 presents the experimental data for the prototype at the design point as stated in Table 1. A comparison

of the dynamic performance shows a good agreement with the calculated values. However, there is a significant disagreement in terms of refrigerating capacity and COP due to increased regenerator losses. This was deduced from an experimental investigation of the viscous losses within the machine, as the pressure drop measured across the regenerators was smaller than predicted. Thus the regenerators must have preferred flow passages for the working gas. Such passages occurred most probably during the assembling procedure in the form of gaps between the regenerator matrix and the cylinder wall. The evaluation of the pressure drop data revealed a gap, which is approximately three times larger than the passages in the matrix for each regenerator. This results in a reduction of the flow through the regenerator matrix of 45% (cold region) and 66% (hot region) or in a reduction of the heat transfer area of approximately 87% for each regenerator. This strongly affects the refrigerating capacity and the COP, and it explains the reduced performance of the prototype. For that reason, the error of the experimental results of the refrigerating capacity and the COP has to be discussed. In order to maintain turbulent flow in the ethanol jacket, the velocity of the ethanol flow cannot be decreased below a certain limit. This results in a maximum temperature difference between entrance and exit of the ethanol of 0.8 °C. The temperature difference was recorded with an error of ± 0.1 °C or $\pm 12.5\%$. Thus the error of the refrigerating capacity was $\pm 12.7\%$ and the COP was measured with an error of $\pm 12.8\%$. In contrast, the error of the remaining data was smaller than $\pm 3\%$.

However, it was pointed out that the prototype was not built to achieve an optimum performance in terms of refrigerating capacity and COP. On the contrary, the operational characteristics of a free-piston Vuilleumier machine were to be investigated in detail. For that reason, the displacer masses, the spring constants, the displacer rod area, the mean pressure and the three operating temperatures were varied. While the masses, the spring constants and the displacer rod area are well suited to

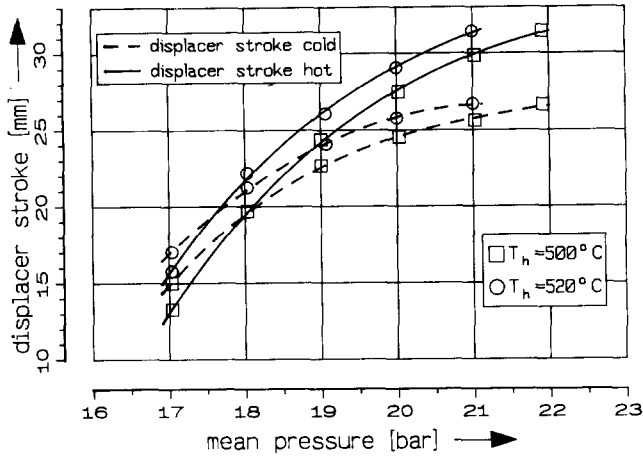


Figure 5 Displacer strokes vs mean pressure: $T_w = 40^\circ\text{C}$, $T_c = 0^\circ\text{C}$
 Figure 5 Course des pistons en fonction de la pression moyenne: $T_w = 40^\circ\text{C}$, $T_c = 0^\circ\text{C}$

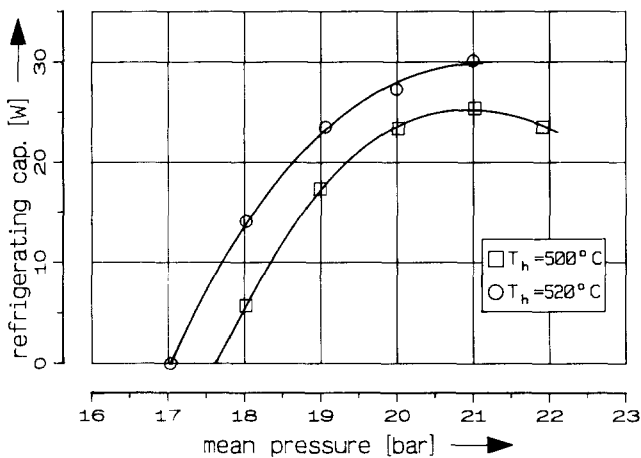


Figure 6 Refrigerating capacity vs mean pressure: $T_w = 40^\circ\text{C}$, $T_c = 0^\circ\text{C}$
 Figure 6 Capacité de refroidissement en fonction de la pression moyenne: $T_w = 40^\circ\text{C}$, $T_c = 0^\circ\text{C}$

adjust the dynamics, as their effect on the oscillation system is obvious, the refrigerating capacity and the COP are mainly affected by the mean pressure and the temperatures. This will be discussed in the following sections.

Mean pressure

The displacer strokes increase with the mean pressure (Figure 5), because the pressure difference across the displacer rod area is increased as well. However, this relationship is not linear, as the increasing density of the working gas causes an increased turbulent pressure drop. The plot of the refrigerating capacity versus the mean pressure has almost the same shape as the plot of the displacer strokes (Figure 6). This implies that the effect of the frequency, the displacer phase and the stroke ratio on the refrigerating capacity is small compared to the displacer strokes. In order to explain this phenomenon, it should be noted that the operating frequency of the oscillation system varies only slightly, if any parameter

is changed, as it is mainly affected by the resonant frequencies of the displacers, which are almost constant. The displacer phase always remained between 80° and 100° during the experiments. The deviations from the ideal phase angle of 90° do not cause serious losses. The effect of the stroke ratio is small, because it varies around its optimum value as well⁹. Thus it can be generally concluded that the relationship between the refrigerating capacity and the dynamics is mainly affected by the displacer strokes.

As shown in Figure 6, the refrigerating capacity reaches a maximum at $p_{\text{mean}} = 21$ bar and $T_h = 500^\circ\text{C}$. In addition to the effect of the displacer strokes, the slope of the curves in Figure 6 decreases even more with increasing pressure, because the increased density of the working gas also causes higher regenerator losses. However, the maximum only occurs owing to the immoderately high regenerator losses in the machine. Nevertheless, it can be concluded that the mean pressure is not suitable to control the refrigerating capacity, as the characteristic line is strongly non-linear.

Hot temperature

In contrast, the refrigerating capacity is a linear function of the hot temperature, which is illustrated in Figure 7. Evidently, the displacer strokes also vary in an almost linear way with the hot temperature. Therefore, the hot temperature seems to be appropriate for controlling the displacer strokes and the refrigerating capacity. Unfortunately, there is a significant decrease of the COP, if the hot temperature is decreased (Figure 7). This results in a reduced performance during operation at partial load. Here, the free-piston machine differs from a crank-driven Vuilleumier machine, where the COP varies only slightly with hot temperature. This phenomenon is due to the strong effect of the displacer strokes on the performance of a free-piston machine, as reduced displacer strokes result in a smaller stroke volume and also in a bigger dead volume, which seriously affects the COP.

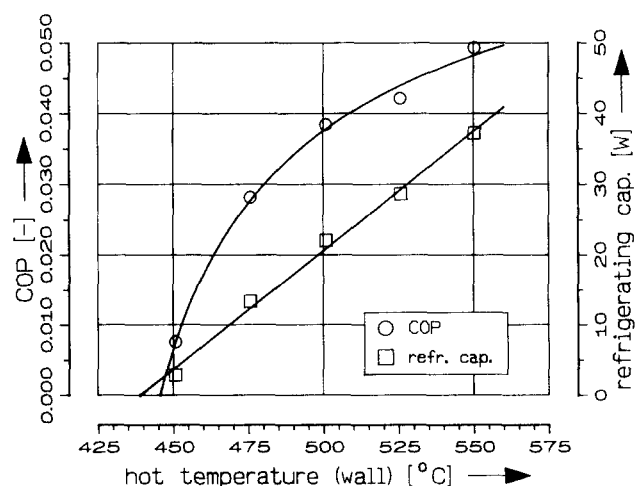


Figure 7 COP and refrigerating capacity vs hot temperature: $T_w = 40^\circ\text{C}$, $T_c = 0^\circ\text{C}$, $p_{\text{mean}} = 20$ bar

Figure 7 COP et capacité de refroidissement en fonction de la température la plus élevée: $T_w = 40^\circ\text{C}$, $T_c = 0^\circ\text{C}$, $p_{\text{mean}} = 20$ bar

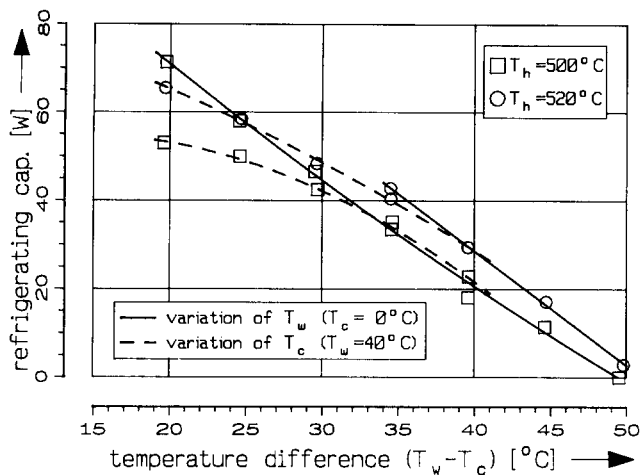


Figure 8 Refrigerating capacity vs $(T_w - T_c)$: $p_{\text{mean}} = 20$ bar
 Figure 8 Capacité de refroidissement en fonction de $(T_w - T_c)$ pour $p_{\text{mean}} = 20$ bar

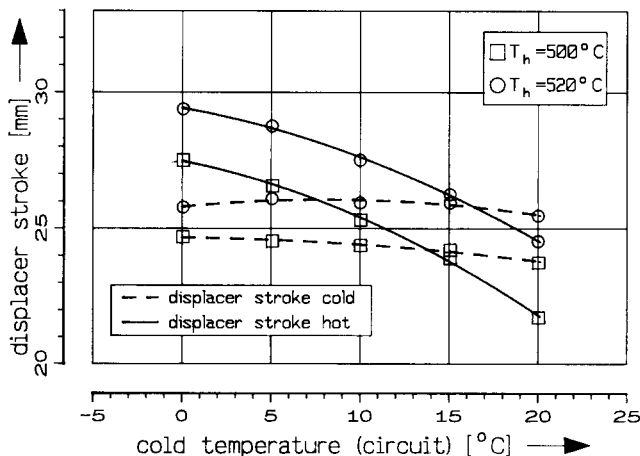


Figure 9 Displacer strokes vs cold temperature: $T_w = 40^\circ\text{C}$, $p_{\text{mean}} = 20$ bar
 Figure 9 Course des pistons en fonction de la plus basse température pour $T_w = 40^\circ\text{C}$ et $p_{\text{mean}} = 20$ bar

Cold and warm temperature

The effects of either the cold or the warm temperature on the performance of a free-piston Vuilleumier machine will be presented in this section. As the refrigerating capacity and the COP are to be discussed, it is useful to examine the effect of the temperature difference $(T_w - T_c)$ instead of the individual temperatures.

In Figure 8, the refrigerating capacity is plotted versus the temperature difference $(T_w - T_c)$. It can be seen that the refrigerating capacity decreases if the temperature difference increases. This is due to increased regenerator and heat conduction losses. However, the curves obtained by the variation of the cold temperature (dashed line) show a smoother slope, especially at low temperature differences. In order to explain this phenomenon, the effect of each temperature on the displacer strokes has to be examined. While the displacer strokes decrease, when the warm temperature is raised, the strokes remain almost constant (cold displacer) or increase (hot displacer), when the cold temperature is decreased (see Figure 9).

Thus the machine partly compensates for higher losses and smaller COPs at low cold temperatures by an increased stroke of the hot displacer. Unfortunately, looking at the experimental data for the prototype, this effect is not significant owing to the high regenerator losses. However, this is an interesting feature, as the refrigerating capacity of a free-piston Vuilleumier machine is less sensitive to changes of the cold temperature than it is for crank-driven or compression-cycle machines.

Figure 10 shows the effect of the temperature difference on the COP. Evidently, the COP decreases if the temperature difference increases, owing to higher regenerator and heat conduction losses. In this case, the slopes of the curves obtained by variation of either the warm or the cold temperature do not vary significantly. The COPs obtained with a constant cold temperature of 0°C are slightly smaller, owing to higher insulation losses at the cold end of the machine. The plot of the ratio $\text{COP}/\text{COP}_{\text{Carnot}}$ versus the temperature difference shows a maximum for 25°C (Figure 11). The maximum can be confirmed theoretically, although in the experiments it occurs at smaller temperature differences owing to the

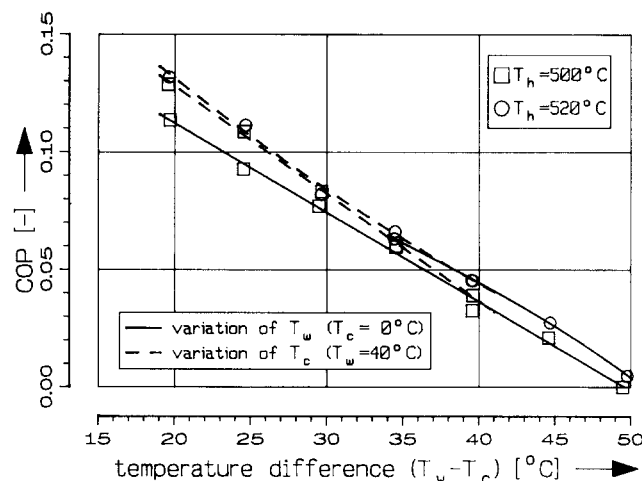


Figure 10 COP vs $(T_w - T_c)$: $p_{\text{mean}} = 20$ bar
 Figure 10 COP en fonction de $(T_w - T_c)$ pour $p_{\text{mean}} = 20$ bar

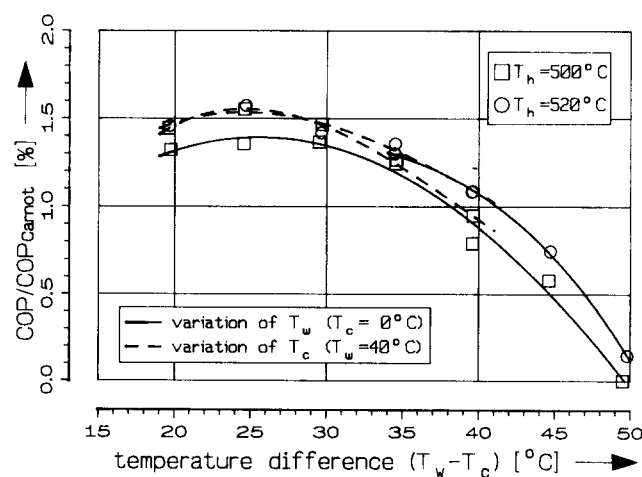


Figure 11 $\text{COP}/\text{COP}_{\text{Carnot}}$ vs $(T_w - T_c)$: $p_{\text{mean}} = 20$ bar
 Figure 11 $\text{COP}/\text{COP}_{\text{Carnot}}$ en fonction de $T_w - T_c$ pour $p_{\text{mean}} = 20$ bar

high regenerator losses of the prototype. Nevertheless, this indicates that a free-piston Vuilleumier machine reaches its maximum COP/COP_{Carnot} for a temperature difference that is suitable for refrigeration or air-conditioning purposes. In contrast, Stirling machines reach their maximum COP/COP_{Carnot} at temperature differences of 100–150°C, which is more useful for cryogenic applications¹⁰.

Conclusion

A free-piston Vuilleumier machine was investigated approximately for the first time. Although the designed refrigerating capacity and COP could not be reached owing to excessive regenerator losses, the interdependence with the mean pressure and the cycle temperatures revealed some interesting features.

The refrigerating capacity is not a linear function of the mean pressure. Instead, the slope of the curves decreases with increasing pressure, as the turbulent pressure drop and the regenerator losses also increase. On the other hand, the refrigerating capacity is a linear function of the hot temperature. The COP also increases with the hot temperature, resulting in a reduced performance during operation at partial load, if the refrigerating capacity is controlled by the hot temperature.

Furthermore, it can be deduced from the experimental data that the cold temperature hardly affects the refrigerating capacity. This is a particular advantage over crank-driven and compression-cycle machines, because the performance of a free-piston Vuilleumier refrigerator is less sensitive to load. Additionally, the COP to

COP_{Carnot} ratio reaches its maximum in the region of the temperature difference between the warm and the cold end, which is commonly used for air-conditioning and refrigerating applications. Thus the free-piston Vuilleumier machine has proved its potential for CFC-free cooling, and further investigation is under way in order to design a machine on a technical scale.

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